(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

[Continued on next page]

(54) Title: HYDRAULIC VALVE ACTUATION SYSTEMS AND METHODS

(57) Abstract: Hydraulic engine valve actuation systems and methods for internal combustion engines. The systems utilize a proportional valve to regulate the flow of a working fluid to and from a hydraulic actuator controlling the engine valve position. The position of the proportional valve is controlled by high speed valves to control various engine valve parameters, including engine valve takeoff and landing velocities. Returning all valves to a known starting position between engine valve events avoids accumulation of errors in proportional valve positioning. Embodiments using spool valves for the high speed valves and the proportional valve, and spring return and hydraulic return for the engine valve, are disclosed. A specially shaped spool in the proportional valve provides enhanced control over the engine valve operation. Various further alternate embodiments are disclosed.

HYDRAULIC VALVE ACTUATION
SYSTEMS AND METHODS

BACKGROUND OF THE INVENTION

1.   Field of the Invention

The present invention relates to the field of hydraulic
valve actuation for internal combustion engines.

2.   Prior Art

At the present time, piston-type internal combustion
engines of interest to the present invention are currently
widely used in automobiles, trucks, buses and various other
mobile and stationary power systems. Such engines include
the common gasoline and diesel engines, as well as similar
engines operating from different fuels such as liquid
propane. These engines commonly utilize intake and exhaust
valves that are spring loaded to the closed position and
which are directly or indirectly opened at appropriate times
by a camshaft driven from the engine crankshaft. In a two-
cycle engine such as a two-cycle diesel engine, the camshaft
will rotate in synchronism with the engine crankshaft, though
in a four-cycle engine, the camshaft is driven through a two-
to-one reduction drive system (gear or chain or belt, etc.)
to rotate at one-half the engine crankshaft speed.

Camshaft actuation of engine valves historically has had
a number of advantages, resulting in its relatively universal
use in such engines for many decades. These advantages
include high reliability, particularly given the current
level of development of such cam actuated valve systems. Cam
actuation is also relatively cost effective, again given the
state of development and quantities in which it is produced.
Cam actuation also has the advantage of allowing shaping the

cam to provide a smooth curve defining valve position versus
camshaft angle. This results in a rather low velocity
takeoff and initial valve opening, as well as a rather low
velocity valve final closing at low engine speeds, resulting
in minimum noise being generated. It also results in faster
valve opening and valve closing at higher engine speeds as
required to maintain the same valve timing throughout the
engine speed operating range.

Cam actuated valve systems also have certain limitations
which are becoming of increasing concern. In particular,
optimal valve timing is not fixed throughout the engine
operating range. For instance, valve timing for maximum
power at one engine speed will not be the same as valve
timing for maximum power at another engine speed.
Accordingly, the classic cam operated valve systems utilize a
compromise valve timing, providing reasonable performance
over a reasonable range of engine operating conditions while
being less than optimal for most, if not at all, these
conditions. Further, valve timing for maximum power at any
engine speed may not be optimal from an engine emissions
standpoint. Optimum valve timing at any given engine speed
may need to be dependent on engine loading, and perhaps other
parameters, such as air temperature, air pressure, engine
temperature, etc.

Recently, mechanisms have been introduced to attempt to
make up for some of the limitations in the fixed timing cam
operated valve systems. These mechanisms include mechanisms
for varying valve timing (but not valve opening duration in
terms of camshaft angle) with engine speed, as well as
mechanisms for also increasing the valve open duration.
However, such mechanisms tend to be complicated, open the
valve a fixed distance under all engine operating speeds and

2

are limited in the number and range of variables for which
valve operation may begin to be optimized.

Recently various hydraulic systems for valve actuation
have been proposed. These systems offer the potential of
more flexible control of valve actuation parameters over the
range of the various engine operating parameters. The
present invention is an improvement on these systems.

BRIEF SUMMARY OF THE INVENTION

Hydraulic engine valve actuation systems and methods for
internal combustion engines. The systems utilize a
proportional valve to regulate the flow of a working fluid to
and from a hydraulic actuator controlling the engine valve
position. The position of the proportional valve is
controlled by high speed valves to control various engine
valve parameters, including engine valve takeoff and landing
velocities. Returning all valves to a known starting
position between engine valve events avoids accumulation of
errors in proportional valve positioning. Embodiments using
spool valves for the high speed valves and the proportional
valve, and spring return and hydraulic return for the engine
valve, are disclosed.

To provide enhanced control over the engine valve
operation, a specially shaped spool in the proportional valve
may be used to shape the flow areas versus spool position.
This allows more gradual restricting of the flow areas versus
spool movement over selected portions of the possible spool
positions, diminishing the effect of small errors in spool
position in such regions without inhibiting the maximum flow
areas when the spool is at its maximum positions.

Various further alternate embodiments are disclosed.

3

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a block diagram of an exemplary configuration of a system in accordance with the present invention.

Figure 2 is a diagram illustrating the general structure and function of the three-way proportional spool valve 24 of Figure 1.

Figure 3 is a perspective view of the spool 38 of the proportional valve of Figure 2.

Figure 4 is an expanded view of an edge of the center land of the spool 38 of Figure 3.

Figures 5 and 6 are graphical representations of the flow area versus spool position provided by the proportional valve 24 between the high pressure rail and the chamber 26 of the valve actuator, and between the chamber 26 and the vent 37, respectively.

Figure 7 is a cross sectional view of an engine valve actuator consisting of two concentric pistons that may be used with the present invention.

Figure 8 is a diagram of an embodiment of the present invention that controls a hydraulically returned engine valve using a closed center 3-way proportional valve.

Figure 9 is a diagram of an embodiment of the present invention that controls a hydraulically returned engine valve using a closed center 4-way proportional valve.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

The present invention is a hydraulic valve operating system for operating one or more intake valves or one or more exhaust valves in a piston-type internal combustion engine, which provides full flexibility in valve timing, valve duration, extent of opening, and valve opening and closing velocity. Operation over the desired range of these and other parameters may be controlled, and more importantly optimized, for all engine operating conditions. Such optimization may also include incrementally adjusting the valve operation based on the valve operation during a previous valve operating cycle. This is achieved by controlling the position of a proportional valve by the use of pilot valves to control the operating parameters of an intake or exhaust valve. In that regard, a reference herein and in the claims to an "intake valve" or an "exhaust valve," unless otherwise made clear by the context in which the phrase is used, shall mean one or more intake valves for a cylinder of an internal combustion engine, or one or more exhaust valves of a cylinder of an internal combustion engine. Exemplary embodiments of this system, sometimes referred to herein as a "two-stage" system, are hereafter described in detail.

First referring to Figure 1, a block diagram of an exemplary configuration of a system in accordance with the present invention may be seen. The system illustrated in Figure 1 may be used to actuate an intake or an exhaust valve. This 2-stage system consists of 2 miniature 2-way digital latching spool valves 20 and 22 coupled to control the position of a 3-way proportional spool valve 24. The proportional spool valve, in turn, controls the flow area into, and out of, a control volume 26. This control volume

5

acts on an actuator 28 to regulate the position of the engine
valve 30.   In this embodiment, a spring return 32 is utilized
for valve closing, though embodiments with hydraulic valve
closing may also be used, as shall be subsequently described.

The 2 miniature 2-way digital latching spool valves 20
and 22 (referred to herein as pilot valves) may preferably be
identical valves, preferably in accordance with the 2 way
valves disclosed in U.S. Patent No. 5,640,987 entitled
Digital Two, Three, and Four Way Solenoid Control Valves,
issued June 24, 1997, the disclosure of which is incorporated
herein by reference.   Such valves are double solenoid, high
speed, magnetically latching spool valves, that as used in
the present invention, are operable between two positions.
The first position couples a first port to a second port for
fluid communication between the two ports, and the second
position blocks fluid communication between the first and
second ports.   While other types of valves could be used,
such as poppet valves, valves generally of the type disclosed
in the above referenced patent are preferred because of their
very high speed for good control, and low energy consumption
because of such capabilities as their magnetic latching, and
the ability to sense completion of actuation, if used, to
minimize heating above the already relatively warm
environment in which they operate.   (See U.S. Patent Nos.
5,720,261 and 5,954,030.)

In the embodiment of the present invention of Figure 1,
valve 20 allows fluid flow from fluid line 34 to a drain line
or reservoir 37 (at a relatively low pressure, such as
atmospheric pressure) when in its first position, and blocks
fluid flow from fluid line 34 to the drain 37 when in its
second position.   Valve 22 allows fluid flow from a low
pressure rail 36 to the fluid line 34 when in its first

6

position, and blocks fluid flow from the low pressure rail 36 to the fluid line 34 when in its second position. Check valve 23 is optional, and is normally closed, as the differential pressure on the check valve normally will not be in a direction to open the valve. Its presence however, will help damp transient pressure fluctuations and recover energy in the pressure fluctuations.

Now referring to Figure 2, a diagram illustrating the general structure and function of the three-way proportional spool valve 24 of Figure 1 may be seen. The proportional spool valve includes a spool 38 within an internal housing 40 which fits within an external housing assembly (not shown) with O-rings in O-ring grooves 42 to separate the regions of ports 1, 2 and 3 from each other and from the ends of the internal housing 40. In that regard, the outer housing assembly, in addition to having the associated fluid connections, also includes internal annual grooves adjacent each of the regions identified as ports 1, 2 and 3 in Figure 2, each to act as a manifold region for the holes through the internal housing 40 for fluid communication with a respective one of the inner regions 44, 46 and 48 in the internal housing 40, respectively. Fluid communication from each of the ports to the associated inner region 44, 46 or 48 is provided in the exemplary embodiment not only by through holes 50, but also by cooperatively disposed orthogonal through holes 52 associated with each of the ports.

As schematically illustrated in Figure 2, the spool 38 is positioned within the internal housing 40 by fluid pressures acting on a piston at the left end of the spool having an effective area A_1 and a piston at the right side of the spool having an effective area of A_2. As specifically

illustrated in Figure 2, the spool 38 is shown in its extreme right position, referred to herein as its first position, as defined by stops on the travel of either the pistons actuating the spool or stops acting on the spool itself. In this position, the spool 38 is blocking fluid communication between ports 3 and 2 and is allowing fluid communication between ports 2 and 1. Obviously, when the spool is at its left-most position, referred to herein as its second position, fluid communication between ports 1 and 2 is blocked and fluid communication between ports 2 and 3 is enabled.

Normally in a spool valve, by way of example in the two miniature, two-way digital latching spool valves 20 and 22 of Figure 1, fluid communication between two adjacent ports will be blocked when the spool is in one position and during the initial motion of the spool toward the other position. However, once the relief on the spool associated with the land in the housing separating the regions coupled to the two adjacent ports starts to bridge the land, a flow area between the regions coupled to the two ports is established, that flow area increasing linearly with further motion of the spool. Because that flow area is a peripheral flow area of the full diameter of the spool, once opening starts, a relatively large flow area between the two ports will be opened with only a relatively small further motion of the spool.

However, in the three-way proportional spool valve 24 (Figure 1), some of the details of which are illustrated in Figure 2, this change in flow area versus spool position is purposely modified to reshape the flow area versus spool position. In the exemplary embodiment, this is accomplished in the manner illustrated in Figures 3 and 4. In that

8

regard, Figure 3 is a perspective view of the spool 38 and
Figure 4 is an expanded view of an edge of the center land of
the spool 38. As may be seen in Figure 3, the center land on
the spool has a plurality of kerfs 54 equally spaced around
each end of the center land, which kerfs begin to open a
controlled flow area with spool position prior to the edge of
the land on the spool reaching the edge of the land on the
internal housing, the normal position for a spool valve flow
area starting to be established.

In addition, as may be seen in Figure 4, small steps are
ground in the center land of the spool of the three-way
proportional spool valve of the exemplary embodiment. Thus,
while the spool has an outer diameter D_0 having a close
sliding fit within the inner diameter of the internal
housing, each end of the center land of the exemplary spool
has additional diameters D_1, D_2 and D_3, where D_3 is less than
D_2, D_2 is less than D_1 and D_1 is less than D_0. This provides
a non-linear variation in flow area versus spool position
during the opening and closing of the fluid communication
between adjacent ports, as illustrated in Figures 5 and 6.
These figures illustrate the flow area between ports 1 and 2,
and ports 2 and 3, respectively, versus the position of the
spool in the three-way proportional spool valve. As may be
seen in Figure 5, when the spool is at the right-most
position, the flow area between ports 1 and 2 is a maximum,
initially decreasing at a relatively high rate for the
initial motion of the spool to the right, then decreasing in
rate for another part of the motion, then decreasing at a
further reduced rate to a substantially zero flow area for
the rest of the spool motion, essentially blocking
communication between ports 1 and 2 when approximately 40% of
the spool motion has been achieved. In comparison, Figure 6

9

shows the flow area between ports 2 and 3, which is a mirror image of Figure 5.

It will be noted from Figures 5 and 6 that in the exemplary embodiment of the present invention, the reduction in flow area on the initial valve closing motion of the spool occurs at a high rate with respect to spool position, decreasing in change in flow rate with an increasing position of the spool until the flow area goes to substantially zero when less than half of the spool motion has been achieved, thereby substantially altering the flow area versus spool position characteristic of a conventional spool valve. Also, because the flow area goes to substantially zero before one-half of the maximum spool travel has been achieved, fluid communication between both ports 1 and 2, and ports 2 and 3, is disabled or blocked when the spool is approximately centered within its travel range. For the specific exemplary embodiment illustrated, the substantial blockage between both ports 1 and 2, and ports 2 and 3, occurs whenever the spool's position is anywhere between approximately 40% of its travel and 60% of its travel. Obviously other shaping of the flow areas, or no shaping may be used if desired, though preferably some shaping will be used to diminish the effect of small errors in spool position in the restricted regions without inhibiting the maximum flow areas when the spool is at its maximum positions.

Referring again to Figure 1, it may be seen that fluid in the low pressure rail 36, which may have a pressure, by way of example, of 20 to 50 bar, is coupled to the right side of the three-way proportional spool valve 24 to act on the area A_2 (Figure 2) of a piston encouraging the spool to its left-most position.

10

Assuming spool valve 22 is open and spool valve 20 is closed, pressure in the low pressure rail 36 is communicated to line 34, and thus acts on area A_1 of the piston actuating the spool of the proportional spool valve (Figures 1 and 2). Because the area A_1 is larger than the area A_2, the spool of the proportional spool valve is forced to its right-most position, coupling port 1 and port 2 to couple chamber 26 to vent, allowing the valve return spring 32 to force the valve 30 to the closed position. Preferably area A_1 is approximately twice area A_2 so that $A_1 - A_2$ A_2.

If the two-way valve 22 is closed and the two-way valve 20 is open, line 34 will be vented to the drain 37, so that the pressure acting on piston area A_1 (Figure 2) of the three-way proportional spool valve will be substantially zero. The pressure acting on area A_2 of the spool valve, however, will be equal to the pressure of the low pressure rail 36, thereby creating an unbalanced force on the spool to force the spool to its left-most position. In this position, port 2 is in fluid communication with port 3, communicating the pressure in the high pressure rail 56 to control volume 26 to force the valve 30 open.

If, by way of example, valve 30 is half open and spool valves 20 and 22 are both closed, then port 2 of the proportional spool valve will be isolated from both ports 1 and 3, so that the fluid in the control volume 26 is trapped, maintaining the valve 30 at its present position. Finally, since the two-way spool valves 20 and 22 are very high speed valves, they may be controlled in such as manner as to rapidly controllably place the spool of the proportional spool valve at any desired location within the extremes of its travel, and thus variably control the flow rate of fluid

11

into or out of the control volume 26. This, in turn, allows
full control of the operating parameters of the valve 30,
such as the extent of opening, the timing and duration of
opening, the velocity profile of the opening and closing of
the valve (which profiles can be different from each other
and/or vary with engine operating conditions), and the final
valve closing velocity with engine rpm. This allows a
relatively low velocity valve closing at low engine rpm for
low noise operation, while still allowing the closing
velocity to be increased with engine rpm, as necessary for
higher engine operating speeds.

The fluid used in the exemplary embodiment in the low
pressure rail, the high pressure rail and passed to drain is
engine operating oil, though other fluids may be used if
desired. Since the flow rates in the control system for
valve 30 will vary with various parameters, such as oil
viscosity, and thus oil temperature, and the pressure of the
low pressure rail and the high pressure rail, operation of
the valve control system of Figure 1 must reasonably
compensate for such variations. As a first order
approximation, these variations may be reasonably modeled so
that the control system as shown in Figure 1 can reasonably
vary operating durations of valves 20 and 22 to at least
approximate the desired profile of the proportional valve
spool position with engine crankshaft angle, given the
existing engine operating parameters (speed, engine load,
fuel temperature, air temperature, engine oil temperature,
atmospheric pressure, etc.).

In the exemplary embodiment, a small Hall effect sensor
58 is positioned adjacent actuator 28 for the valve 30 so as
to provide a feedback signal to the controller. Thus valve
motion during a valve operating cycle may be monitored and

12

used to control the operation of the valves 20 and 22 for
that valve operating cycle, and/or to make corrections in the
next valve operating cycle to more accurately achieve optimum
valve operation for that valve operating cycle. In that
regard, more optimum operation may be determined in any of
various ways, including better compliance to a predetermined
valve position profile versus engine crank angle as
predetermined for the then existing engine operating
conditions and ambient conditions, or as determined by the
effect of incremental changes on one or more engine
performance characteristics for the change in valve operation
just made, or a combination of both.

In the event two (or more) valves are being actuated in
unison by a single proportional valve 24, a sensor such as a
position sensor (Hall effect sensor or other position sensor)
may be used on only one of the valves, or on both valves, the
sum of the signals providing a better average indication of
the position profile of the two valves and the difference in
the signals providing fault detection, such as a sticky
valve. While a position sensor(s) is preferred, other types
of sensors could be used, such as a velocity sensor, as the
integration times to convert to position are short. In that
regard, at the end of each valve operating cycle, the control
valve 22 is actuated to couple line 34 to the low pressure
rail 36 and control valve 20 is actuated to decouple line 34
from the drain 37 to bring the spool 38 to the stop at the
position shown schematically in Figure 1. This provides
predetermined spool and pilot valve starting points for each
valve operating cycle so that errors in the spool valve
position do not accumulate, valve operating cycle to valve
operating cycle. If desired, a sensor may also be used to

sense the position of the proportional spool valve spool 38, though this is not preferred.

Thus the two miniature latching valves 20 and 22 (sometimes referred to herein as pilot valves) control the position of the proportional valve 24. Specifically, the supply pilot valve 22 allows fluid to flow between a low-pressure rail 32 (approximately 20-50 bar) and a first piston used to move the proportional 3-way valve. The vent pilot valve 20 will allow fluid to flow from the piston to a vent at atmospheric pressure. Using these pilot valves, the position of the proportional valve can be changed quickly and accurately. The position of the proportional valve can be infinitely varied throughout 3 flow states noted in Figures 5 and 6, namely:

> State 1: The high pressure fluid from the high pressure rail 56 (approximately 100-240 bar) is allowed to flow from the high pressure rail to a control volume 26 above the engine valve actuation piston.
> State 2: The spool 38 of the proportional valve is centered between its hard stops, trapping fluid in the control volume above the engine valve actuation piston and creating a hydraulic lock.
> State 3: The fluid in the control volume 26 above the engine valve actuation piston is vented to atmospheric pressure.

As the proportional valve moves from state 2 to state 3, the area through which high-pressure fluid from the high pressure rail 56 can flow into the control volume 26 above the engine valve actuation piston increases nonlinearly. (See Figure 6). Similarly, as the proportional valve moves from state 2 to state 1, the area through which fluid can flow out of the control volume 26 above the engine valve actuation

piston to drain 37 increases nonlinearly (See Figure 5).
Thus the geometry of the proportional spool valve has been
designed with regions of high and low gain. The low gain
region provides fine control for take-off and seating
velocities, while the high gain region provides the large
flow area required to achieve maximum engine valve
velocities. This facilitates more accurate control of the
engine valve during seating and take-off. These areas need
more accuracy so that proper seating velocities and valve
overlap are achieved throughout the full range of engine
speed and temperature.

To better describe the function of the exemplary
hydraulic system, the following description traces the system
through a complete engine valve operating cycle, mimicking
results from a nodal hydraulics simulation. The specific
simulation model used 100°C 0W30 synthetic motor oil at an
engine speed of 6000 rpm, though simulations at lower engine
speeds have also been run.

An exemplary valve event may be described as follows.
Initially the supply pilot valve 22 is open and the vent
pilot valve 20 is closed (as illustrated in Figure 1). This
keeps the proportional valve spool in the venting (right-
most) position (State 3, Figures 5 & 6). Specifically, the
flow area between engine valve actuation piston control
volume 26 and vent 37 is at a maximum (state 3, Figure 6) and
the area between engine valve actuation piston control volume
and the high-pressure rail is closed (state 3, Figure 5). As
a result, the engine valve is forced closed against its seat
by the return spring 32.

To initiate valve opening, the supply pilot valve 22 is
opened and the vent pilot valve 20 is closed. This allows
fluid to flow from the control volume of the proportional

15

spool valve to vent. As a result, the proportional spool
begins to move from state 3. The vent pilot valve 20 is left
open long enough for the proportional spool to pass through
state 2 and into state 1. However, the proportional valve is
only allowed to travel until just a small flow area in the
low gain region of state 1 is open between the high-pressure
rail (Figure 5) and engine valve actuation piston control
volume 26. This results in a slow take-off of the engine
valve. The speed of this take-off will vary depending on
where the proportional valve is stopped. Then the vent pilot
valve 20 is opened once again so that the proportional spool
moves to a position that opens a larger flow area between the
high pressure rail and the engine valve actuation piston's
control volume. This results in a rapid opening of the
engine valve after the initial slow takeoff.

The engine valve now must stop at the desired lift, in
this particular example, 11mm. To do this, the proportional
spool will be moved to state 2 in which the control volume
above the engine valve is hydraulically locked. This is
achieved by closing the vent pilot valve 20 and opening the
supply pilot valve 22 for the required amount of time. The
engine valve will stay in this position until it is commanded
to return. At this point the kinetic energy in the engine
valve is fully converted into potential energy of the fluid
in the control volume and the engine valve return spring.
This trade off between kinetic and potential energy occurs
several times while the control volume is hydraulically
locked, which can result in a slight oscillation of the
engine valve position. To reduce this effect and to recover
some of the kinetic energy in the proportional valve spool, a
check valve could also be placed between the control volume
26 of the engine valve actuator and the high-pressure rail 56

in order to damp out any high pressure spikes that may occur during operation.

Next, the supply pilot valve 22 will be opened again long enough to move the proportional valve to the high gain region of state 3, and then closed, at least before vent pilot valve 20 is again opened. To reiterate, at this point the flow area between the engine valve control volume 26 and vent 37 is a maximum. Therefore, the engine valve will accelerate very quickly toward its seat via the stored energy in the valve spring.

In order to seat the valve at the desired velocity, the flow area that connects the engine valve control volume 26 and vent 37 must be restricted. This can be achieved by once again opening the vent pilot valve 20 for a short period to reposition the proportional valve to a low gain in state 3. This seating velocity will change depending on where the proportional valve is stopped in this region.

This completes one engine valve cycle. In order to prepare the system for the next event, all components will be repositioned to their initial conditions. The only component that is out of position is the proportional valve. The supply pilot valve 22 is again opened, returning the proportional valve to a position of maximum flow area in state 3. This reestablishes a reference point at the beginning of each valve event, so that errors in proportional valve positioning do not accumulate, one valve cycle to another. In this way, the seating velocities desired at different engine speeds, loads and temperatures can be achieved by changing the position at which the proportional spool dwells. This can be facilitated further by varying the pressure in the low-pressure rail if desired, thus accomplishing finer control of the proportional spool valve.

17

In a simulation of the system described above, an engine valve actuator consisting of two concentric pistons was used, as illustrated in Figure 7. Instead of using one actuator with a relatively large area exposed to pressure to drive the engine valve through its entire stroke, the large piston (boost piston 60) is used only initially to achieve peak velocities before reaching a mechanical stop, while the remainder of the stroke is accomplished using a smaller telescoping piston (drive piston 62). Specifically, when the engine valve, particularly an exhaust valve, initiates lift from its seat, in-cylinder pressure remains substantial. In addition, maximum engine valve acceleration is also required at this time. As a result, a greater force is needed to actuate the engine valve through the beginning of its stroke while a much lower force is required for the remainder of the stroke. The present invention system does not rely on using the two concentric piston design, as it will also function if just one actuator is used. However, the two concentric piston design requires less fluid from the high pressure rail for each valve cycle, and thus requires less energy for valve operation.

One can also use a hydraulically returned engine valve as opposed to a spring returned valve, as illustrated in Figure 8. In this embodiment, the valve actuator comprises a piston 64 having a cross-sectional area A_1 on a piston rod 66 having a cross-sectional area A_2. Chamber 68 is permanently coupled to the high pressure rail, and chamber 70 is switchable by the proportional valve between the high pressure rail and the vent. Consequently, the maximum valve opening force is equal to the pressure of the high pressure rail times A_2 and the maximum valve closing force is equal to the pressure of the high pressure rail times A_1-A_2. Because

of the functional relationship between spring force and
stroke, one can achieve essentially the same valve dynamics
as a hydraulically returned valve with a smaller diameter
actuator.

With a return spring, the spring closing force is at a
minimum when one desires a large opening force for maximum
acceleration against peak cylinder pressure. With hydraulic
return, the closing force of a hydraulically returned engine
valve is constant and therefore will be higher than that of a
spring when the valve is seated. Therefore, the force
characteristic of a mechanical spring is more desirable for
returning the engine valves than a single piston return
mechanism.

Instead of using a closed center 3-way proportional
valve, the hydraulically returned system can also be
constructed using a closed center 4-way proportional valve
(Figure 9). Like the closed center 3-way proportional valve,
its position can also be infinitely varied throughout 3 flow
states.

    State 1: The high pressure fluid is allowed to flow
    from the high pressure rail to a control volume 70 above
    the engine valve actuation piston 64 while the fluid in
    chamber 68 acting below the engine valve actuation
    piston 64 is vented to tank.

    State 2: The proportional valve is centered between its
    hard stops, trapping fluid in the control volume 70
    above the engine valve actuation piston and in the
    control volume 68 below the engine valve actuation
    piston, thus creating a hydraulic lock.

    State 3: The fluid in the control volume 70 above the
    engine valve actuation piston is vented to atmospheric
    pressure while high pressure fluid is allowed to flow

from the high pressure rail to the control volume 68
below the engine valve actuation piston 64.

As the proportional valve moves from state 2 to state 1
the area through which high-pressure fluid can flow into the
control volume 70 above the engine valve actuation piston 64
increases nonlinearly (similar to Figures 5 & 6). At the
same time the flow area between the fluid below the engine
valve actuation piston 64 and tank increases nonlinearly.
Similarly, as the proportional valve moves from state 2 to
state 3, the area through which fluid can flow out of the
control volume 70 above the engine valve actuation piston 64
to tank increases nonlinearly. At the same time the flow
area between the fluid below the engine valve actuation
piston 64 and the high-pressure rail increases nonlinearly.

In all the systems described, the proportional valve
uses hydraulic force to oppose the pressure in its control
volume. Alternatively, the proportional valve can also use a
spring to supply part or all of the opposing force.

Any of the 3-way proportional valve systems can take
advantage of the recovery systems known in the art. The low-
pressure rail used for actuating the 3 or 4 way proportional
valve can be used for the low-pressure source of the recovery
system if that system is implemented.

The present invention has many advantages for both
diesel and gasoline engines, as well as similar engines
powered with alternate fuels. These advantages include:
   Infinitely variable engine valve timing for both opening
and closing times.
   Infinitely variable engine valve lift from the valves seat
to its maximum lift position.
   Infinitely variable valve open and/or close time duration.

The proportional 3 or 4 way valve has low gain flow regions for fine control at valve take-off and seating. It also has high gain flow regions for maximum flow allowing increased speed of the engine valve so that airflow into the engine cylinders can be maximized.

The system can allow the engine valve profile to be non-symmetric.

The system is capable of an infinitely varying the slew rate of the engine valve independent of rail pressure.

The system does not require a slow take-off and landing. Specifically, the valve can begin opening with maximum acceleration or seat at maximum velocity if desired.

The system does not need a lash adjustment system, specifically:

> the system is unaffected and can compensate for the growth of engine components (specifically valve train components) due to thermal expansion,

> the system is unaffected and can compensate for engine valve recession due to wear of the valve seat and the engine valve, and

> the system is unaffected and can compensate for tolerance stack up between valve train components resulting from initial assembly and manufacturing tolerances.

The system can compensate for varying working fluid viscosity due to temperature, age, etc.

The system can optimize the amount time at which air is metered into the engine cylinder thus optimizing the combustion event at the full spectrum of engine operating conditions resulting in:

maximum power,

lower emissions,

reduced emissions by controlling fuel/air mixing,

reduced heat rejection by reduction of unnecessary in cylinder air motion, and

high BMEP combustion schemes to improve catalyst light-off, reduce startup emissions.

The system can be operated in such a way that engine braking will result, specifically by shutting off the injector during braking and opening the exhaust valve at the top of the compression stroke to dissipate the compression energy.

The engine cycle can be varied to allow for:

2 stroke operation.

Multiple stroke operation (such as, by way of example 2-stroke to 4-stroke, 4-stroke to 6-stroke or 8-stroke operation, etc.) by eliminating one or more pairs of strokes from the normal engine operating cycle, with the valves being controlled during these pairs of strokes for minimum energy loss and/or other considerations.

The system can allow for internal exhaust gas recirculation (EGR). As a result the EGR valve can be removed.

Variable compression ratio.

Miller cycle operation - Maximum cylinder pressure control with high expansion ratio for maximum thermodynamic efficiency.

Atkinson cycle operation.

Reduced heat rejection by reduction of unnecessary in cylinder air motion.

High BMEP combustion schemes to improve Catalyst
light-off, reduce startup emissions.

Improved Cranking and Cold Start.

Reduced white smoke and diesel "fuel" smell during
startup/cold temperature idle/high altitude operation

High altitude compensation.

Variable torque curves to better fit duty/drive
cycle of vehicle.

Increased torque at low speeds for better
driveability, potential vehicle fuel economy
improvements.

The system will operate more efficiently with a
sequentially apportioned pump.

The low-pressure rail can be replaced with an
accumulator that is supplied by the return flow of the engine
valve actuator.

Because the engine valve motion can be varied so that
air can be throttled at the engine valve, the throttle body
can be eliminated.

Operation of the turbo can be optimized at all engine
operating conditions.

Cylinder deactivation for improved vehicle fuel economy.

This 2-stage system has the capability of satisfactorily
controlling engine valves at very high engine speeds (from
idle speeds to 10,000 RPM). In addition, the critical
regions of valve take-off and seating can be controlled with
accuracy and precision while providing the features of
infinitely variable valve timing, duration and lift. The
system also has the capability of significantly increasing
the amount of air that can be supplied to an engine's

23

cylinders throughout the full range of engine speed by adjusting valve timing and duration to maximize the dynamic effects of flow into and out of the combustion chamber at all engine speeds.

While an exemplary embodiment and various alternate embodiments of the present invention have been disclosed herein, it will be obvious to those skilled in the art that various changes in form and detail may be made therein without departing from the spirit and scope of the invention.
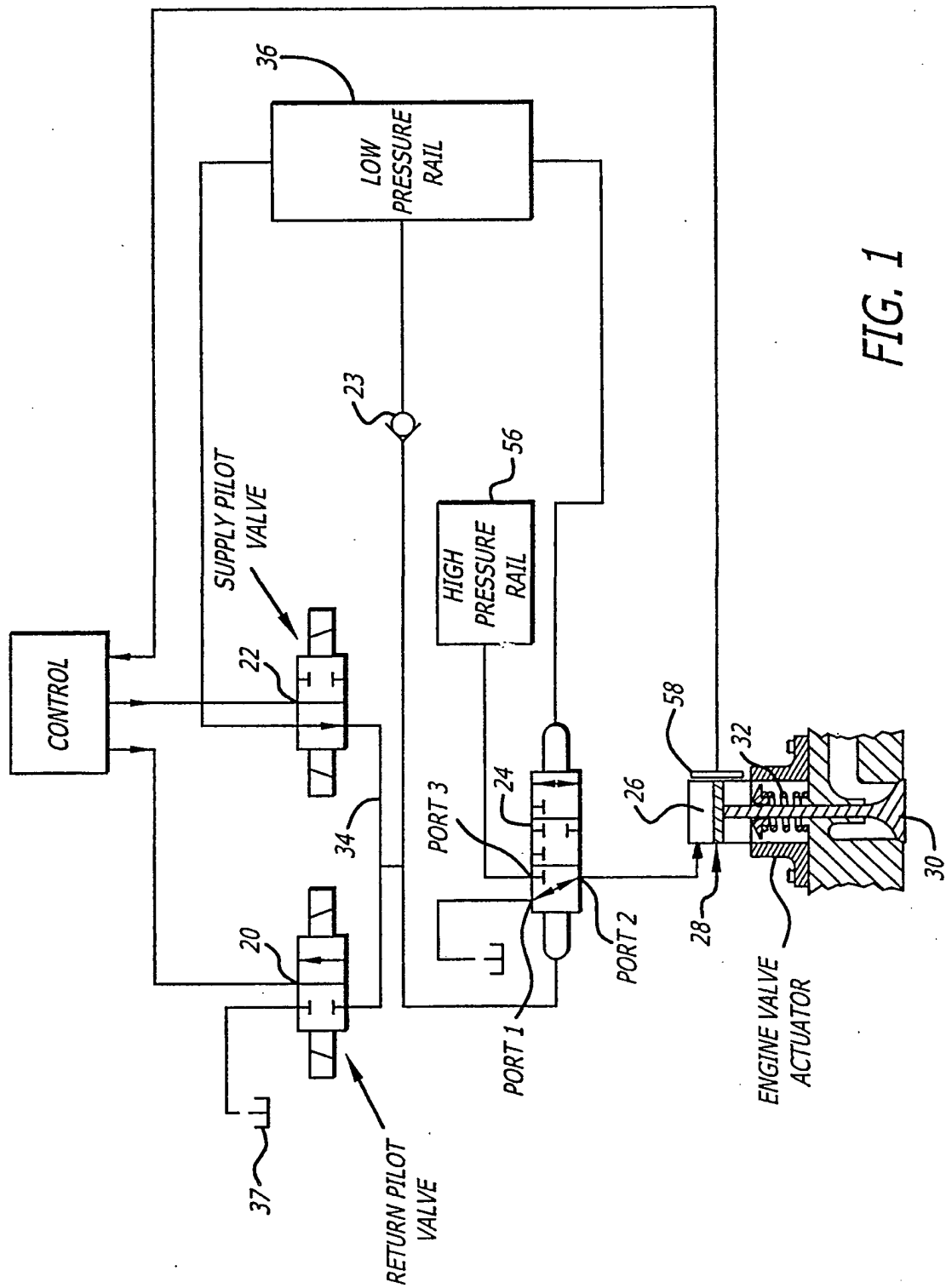
CLAIMS

What is claimed is:

1.   Apparatus for opening an engine valve comprising:

a hydraulic actuator disposed with respect to the valve to encourage the valve toward a valve open position by the pressure of a fluid in the hydraulic actuator;

a proportional spool valve having a spool hydraulically moveable between a first position coupling a source of fluid under a first pressure to the hydraulic actuator and a second position coupling the hydraulic actuator to a reservoir of fluid under a second pressure, the second pressure being less than the first pressure;

electrically controlled valving hydraulically controlling the position of the spool between the first and second positions; and,

a valve return returning the valve to a closed position.

2.   The apparatus of claim 1 wherein the spool valve has a third position between the first and second positions blocking the source of fluid under a first pressure from the hydraulic actuator blocking the hydraulic actuator from the reservoir of fluid under the second pressure.

3.   The apparatus of claim 1 wherein the second pressure is atmospheric pressure.

4.   The apparatus of claim 1 wherein the spool is stepped in diameter to shape the area versus spool position for flow between the source of fluid under the first pressure and the hydraulic actuator, and the hydraulic actuator to the reservoir of fluid under the second pressure.

5.   The apparatus of claim 1 wherein the valving comprises double solenoid latching spool valves.

6.   A method of opening an engine valve comprising:

providing a hydraulic actuator disposed with respect to the valve to encourage the valve toward a valve open position by the pressure of a fluid in the hydraulic actuator;

coupling the hydraulic actuator to a proportional spool valve having a spool hydraulically moveable between a first position coupling a source of fluid under a first pressure to the hydraulic actuator and a second position coupling the hydraulic actuator to a reservoir of fluid under a second pressure, the second pressure being less than the first pressure; and,

hydraulically controlling the position of the spool between the first and second positions by electrically controlled valving.
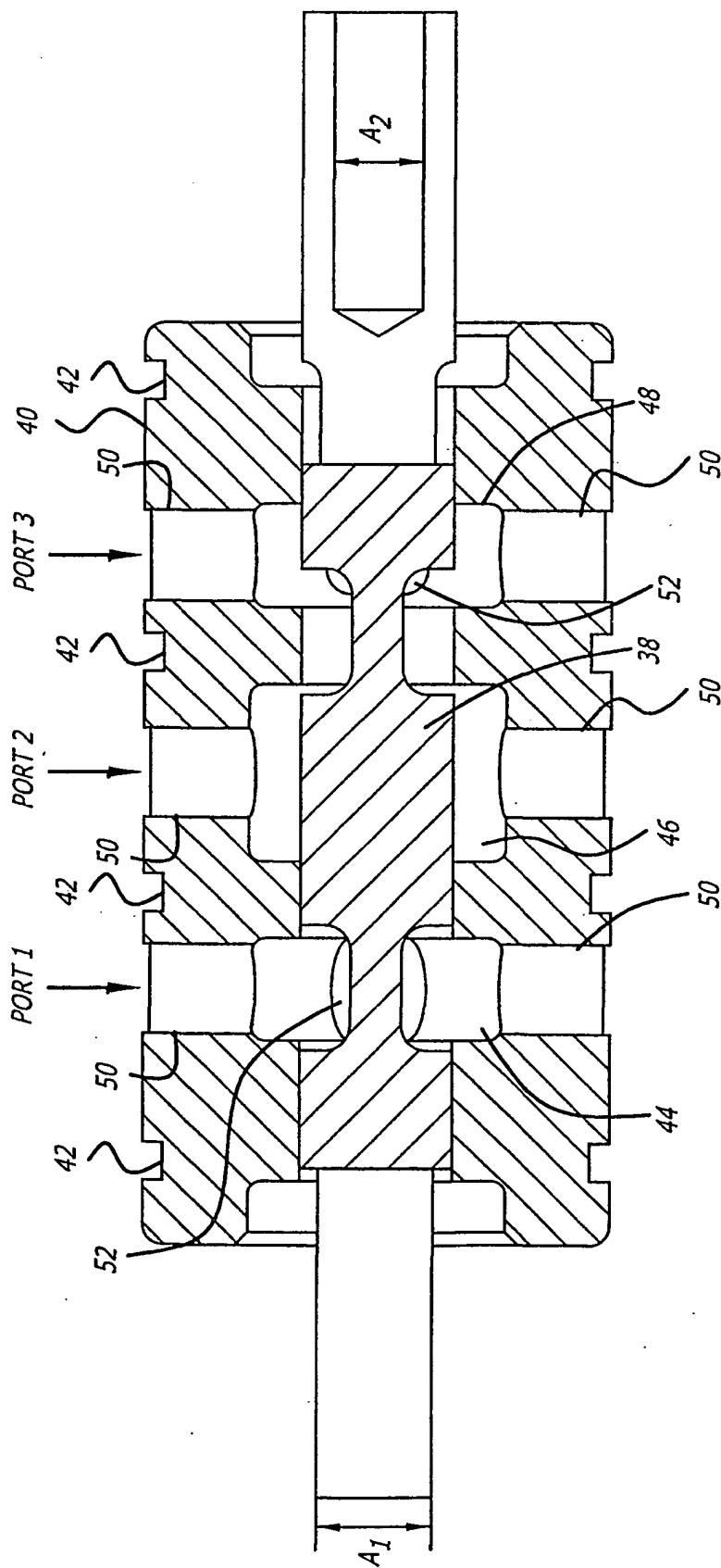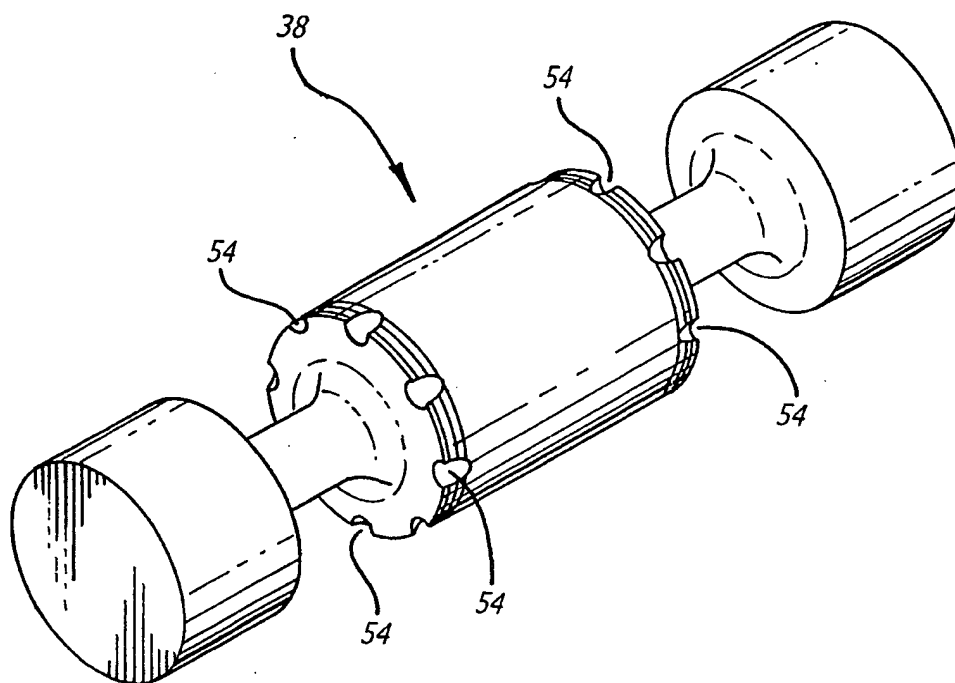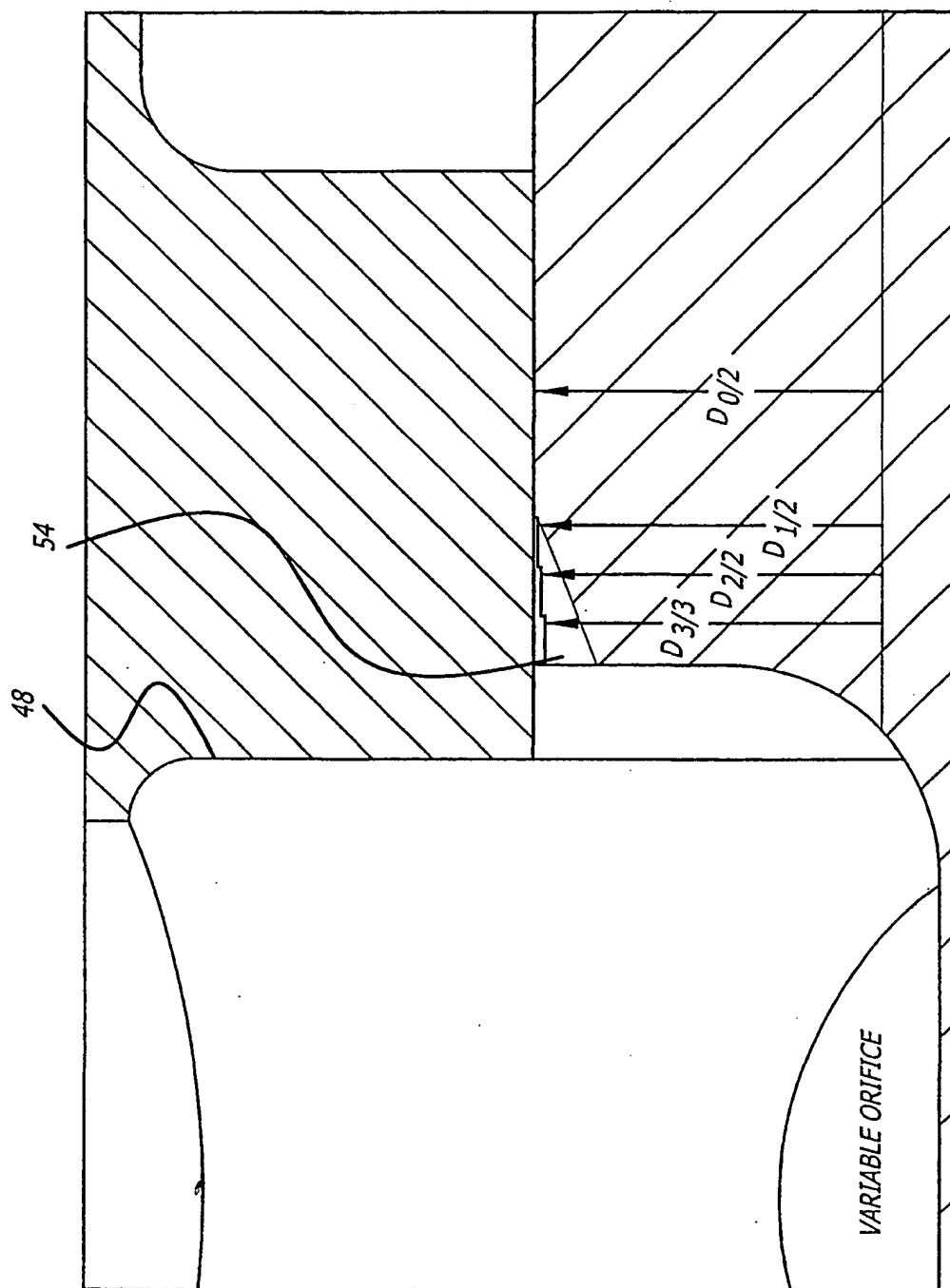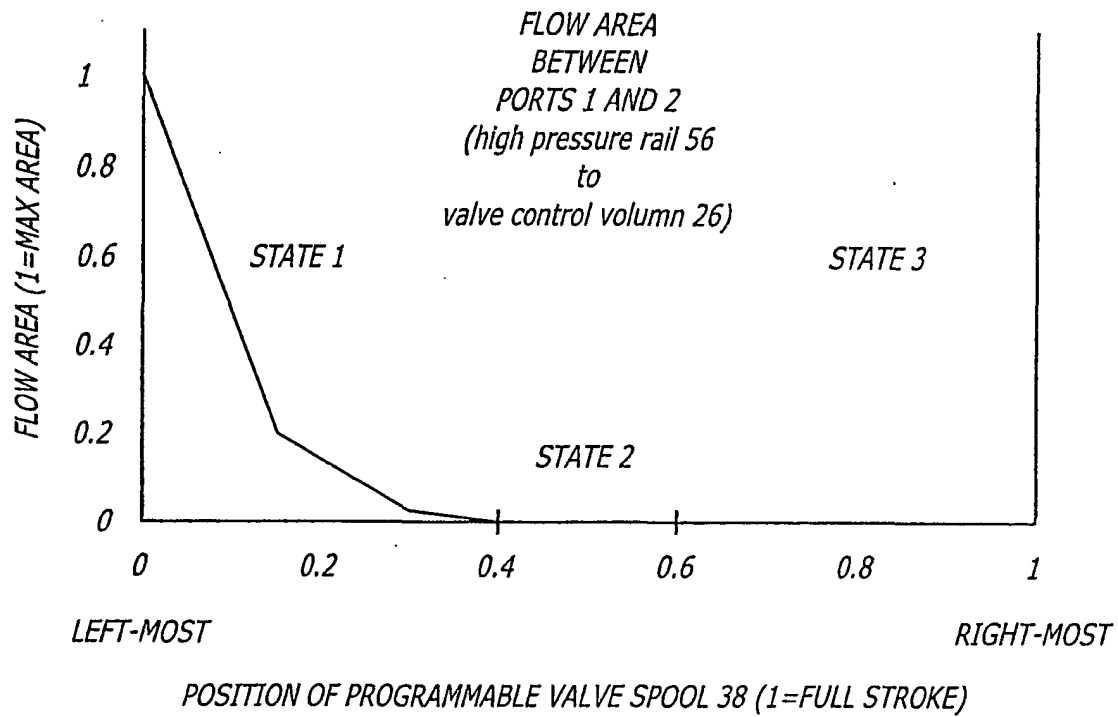
FIG. 1

FIG. 2

FIG. 3

FIG. 4

FIG. 5
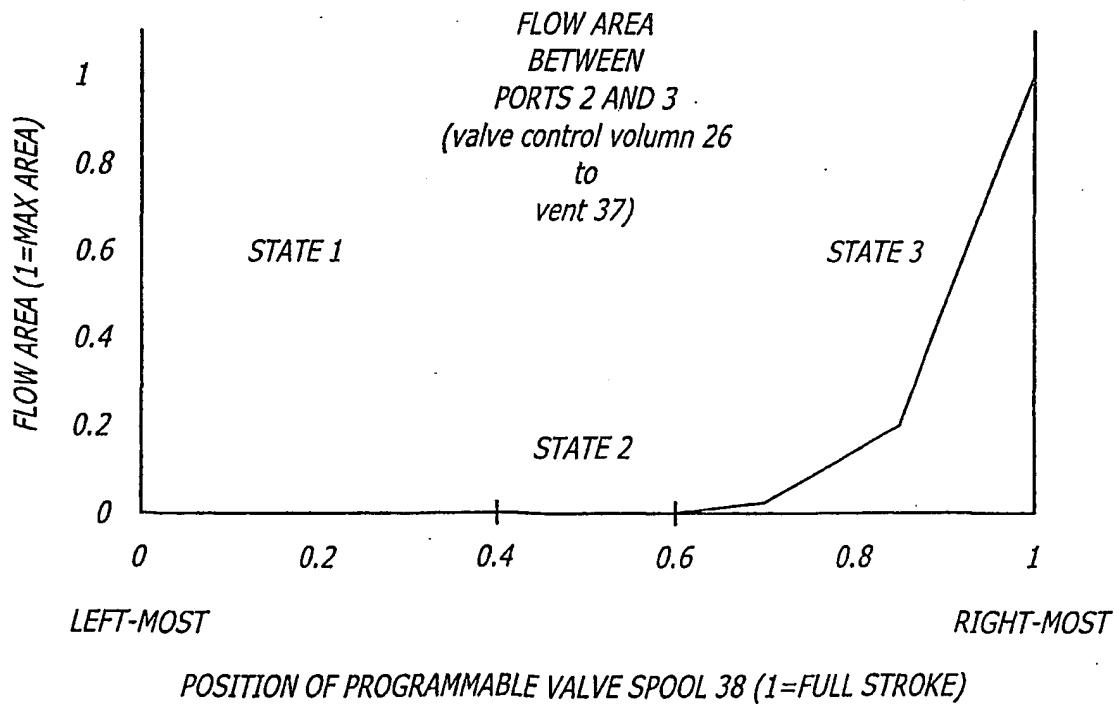
FIG. 6
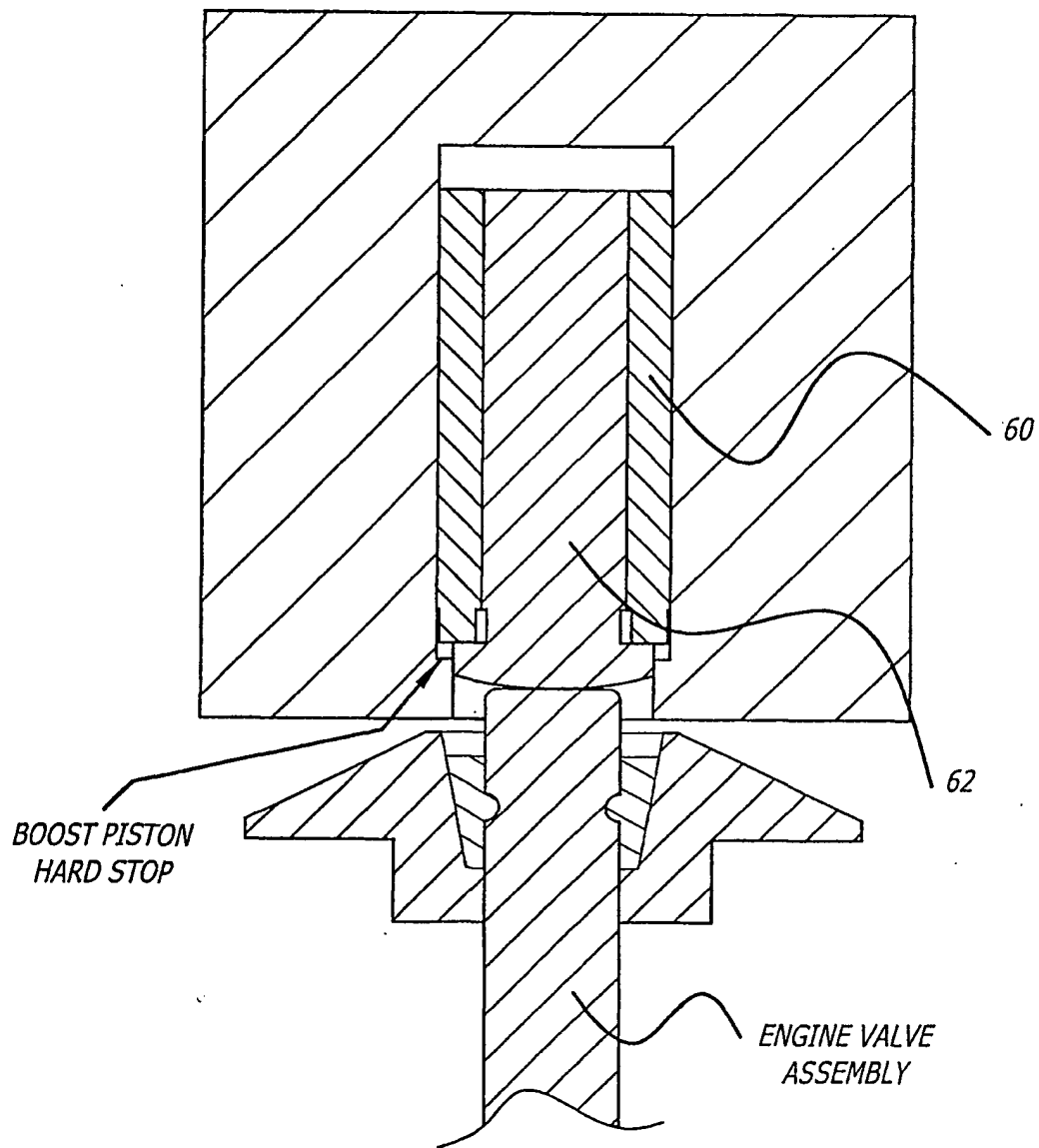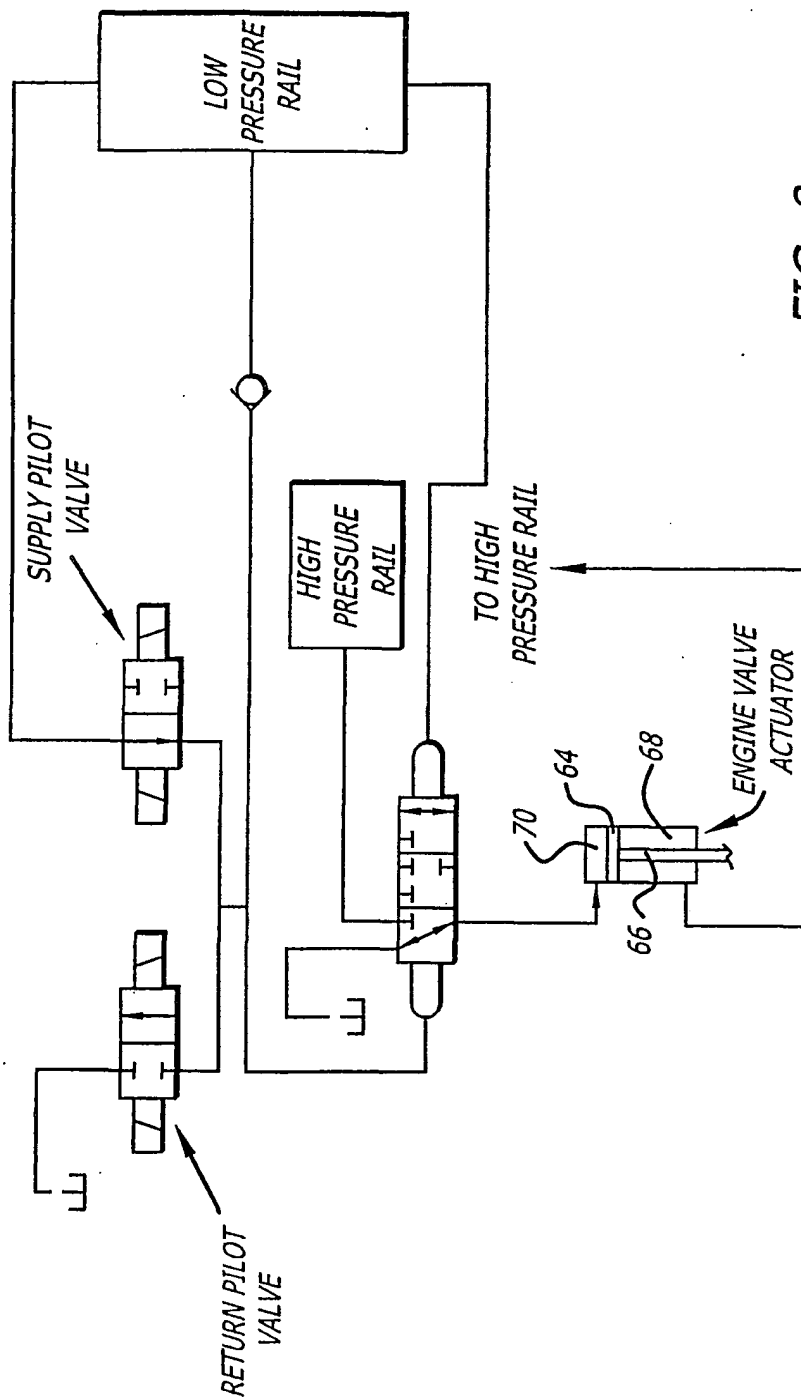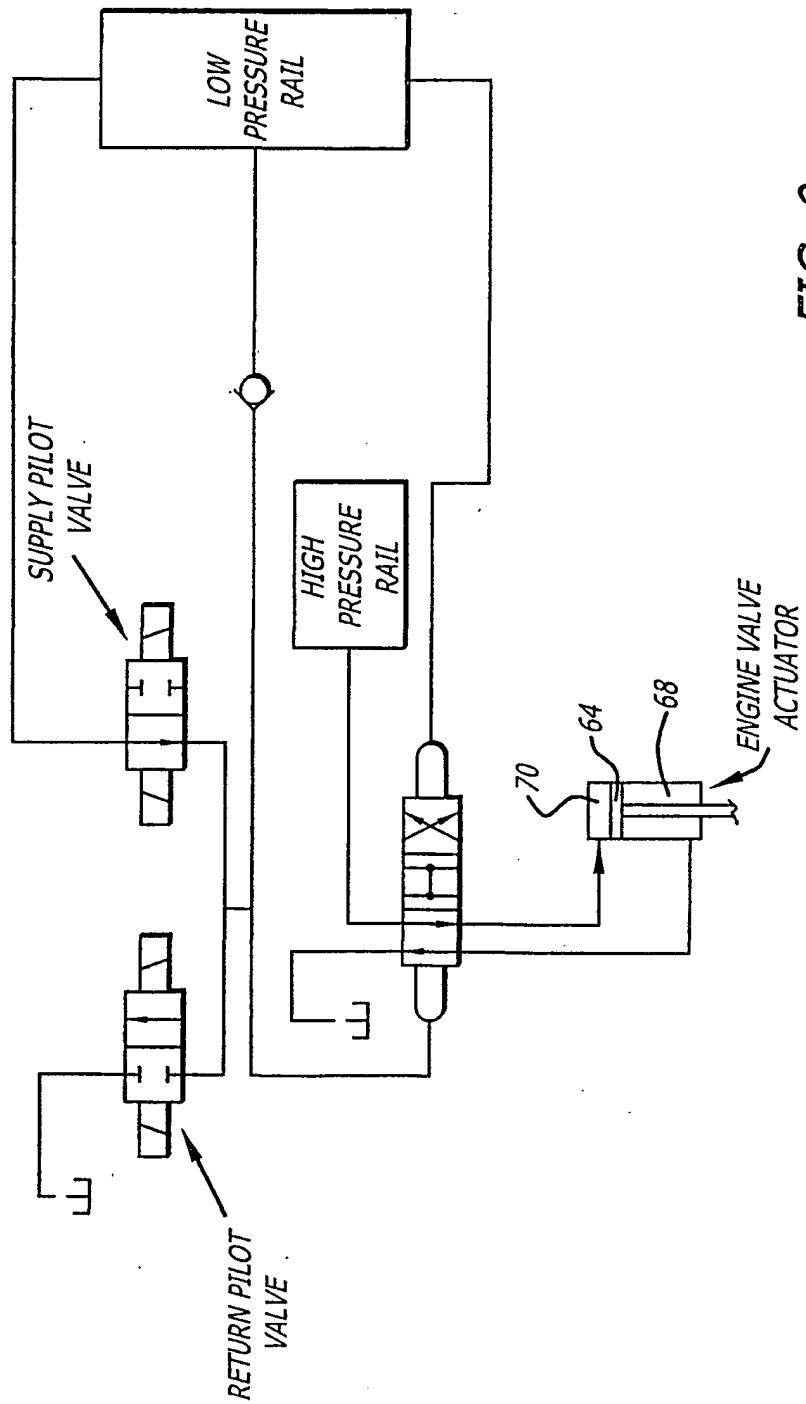
60

62

BOOST PISTON
HARD STOP

ENGINE VALVE
ASSEMBLY

FIG. 7

FIG. 8

FIG. 9

(51) International Patent Classification[7]: F01L 9/02

(21) International Application Number: PCT/US01/46686

(22) International Filing Date:
30 November 2001 (30.11.2001)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
09/729,487   4 December 2000 (04.12.2000)   US

(71) Applicant (for all designated States except US): STUR-MAN INDUSTRIES, INC. [US/US]; One Innovation Way, Woodland Park, CO 80863 (US).

(72) Inventors; and

(75) Inventors/Applicants (for US only): TURNER, Christopher, Wayne [US/US]; 5255 Augusta Trail, Fort Collins, CO 80528 (US). RAIMAO, Miguel, Angelo [US/US]; 335 Via Linda Vista, Manitou Springs, CO 80829 (US). BABBITT, Guy, Robert [US/US]; 822 North Weber, Colorado Springs, CO 80903 (US).

(74) Agents: BLAKELY, Roger, W. et al.; Blakely, Sokoloff, Taylor & Zafman, 12400 Wilshire Boulevard, 7th floor, Los Angeles, CA 90025-1026 (US).

(81) Designated States (national): AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NO, NZ, OM, PH, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, US, UZ, VN, YU, ZA, ZM, ZW.

(84) Designated States (regional): ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Published:
—   with international search report

[Continued on next page]

(54) Title: HYDRAULIC VALVE ACTUATION SYSTEMS AND METHODS

(57) Abstract: Hydraulic engine valve actuation systems and methods for internal combustion engines. The systems utilize a proportional valve to regulate the flow of a working fluid to and from a hydraulic actuator controlling the engine valve position. The position of the proportional valve is controlled by high speed valves to control various engine valve parameters, including engine valve takeoff and landing velocities. Returning all valves to a known starting position between engine valve events avoids accumulation of errors in proportional valve positioning. Embodiments using spool valves for the high speed valves and the proportional valve, and spring return and hydraulic return for the engine valve, are disclosed. A specially shaped spool in the proportional valve provides enhanced control over the engine valve operation. Various further alternate embodiments are disclosed.

(88) Date of publication of the international search report:
16 January 2003

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

INTERNATIONAL SEARCH REPORT

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7   F01L9/02

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7   F01L   F15B   F16K

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5 881 689 A (HOCHHOLZER TIMO) 16 March 1999 (1999-03-16) column 7, line 35 –column 8, line 12; figure 3A	1-3,6
X	EP 0 550 925 A (PHILIPS NV) 14 July 1993 (1993-07-14) claims 1,2; figure 1	1,6
X	DE 12 92 493 B (FRISCH GEB KG EISENWERK) 10 April 1969 (1969-04-10) the whole document	1-3
A	US 5 193 584 A (WATTS THOMAS A) 16 March 1993 (1993-03-16) column 1, line 7 –column 1, line 22; figures 1-3	4

—/—

| X | Further documents are listed in the continuation of box C. | X | Patent family members are listed in annex. |

° Special categories of cited documents :

'A' document defining the general state of the art which is not considered to be of particular relevance

'E' earlier document but published on or after the international filing date

'L' document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

'O' document referring to an oral disclosure, use, exhibition or other means

'P' document published prior to the international filing date but later than the priority date claimed

'T' later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

'X' document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

'Y' document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such docu- ments, such combination being obvious to a person skilled in the art.

'&' document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
18 July 2002	02/08/2002

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Clot, P

International Application No

PCT/US 01/46686

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication,where appropriate, of the relevant passages	Relevant to claim No.
A	US 5 640 987 A (STURMAN ODED E) 24 June 1997 (1997-06-24) cited in the application claims 1,7	5

Patent document cited in search report		Publication date	Patent family member(s)		Publication date
US 5881689	A	16-03-1999	DE	19543080 A1	22-05-1997
			JP	9151715 A	10-06-1997
EP 0550925	A	14-07-1993	US	5248123 A	28-09-1993
			EP	0550925 A2	14-07-1993
			JP	5240011 A	17-09-1993
DE 1292493	B	10-04-1969	NONE		
US 5193584	A	16-03-1993	DE	4304320 A1	26-08-1993
			JP	5288279 A	02-11-1993
US 5640987	A	24-06-1997	AU	2241095 A	30-10-1995
			DE	19580307 T0	17-04-1997
			GB	2301655 A ,B	11-12-1996
			HK	1009345 A1	28-05-1999
			JP	10504873 T	12-05-1998
			WO	9527865 A1	19-10-1995
			US	5598871 A	04-02-1997
			US	6308690 B1	30-10-2001